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Exergy Analysis for Greener Gas Turbine Engine Arrangements

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Abstract—Exergy analysis is a method that uses the conservation of mass and conservation of energy principles together with the second law of thermodynamics for the analysis, design, and improvement of energy and other systems. The exergy method is a useful tool for furthering the goal of more efficient energy-resource use, for it enables the locations, types, and magnitudes of wastes and losses to be identified and meaningful efficiencies to be determined. The exergy analysis of two-shaft gas turbine arrangements is presented and discussed in this paper. Two configurations (in parallel and series free turbine) are presented here and analyzed separately to identify and quantify the energy and exergy losses. Comparison between the two configurations is presented in terms of work output, efficiency, SFC, exergy destruction, and second-law efficiency for the design conditions. The percentage ratio of the exergy destruction in the individual components to total exergy destruction was found maximum in the combustion chambers (above 90%). The second-law efficiency of series configuration is found to be higher than parallel.

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1. INTRODUCTION

Exergy analysis is a method that uses the conservation of mass and conservation of energy principles together with the second law of thermodynamics for the analysis, design, and improvement of energy and other systems. The exergy method is a useful tool for furthering the goal of more efficient energy-resource use, for it enables the locations, types, and magnitudes of wastes and losses to be identified and meaningful efficiencies to be determined [1].

Exergy analysis approach involves the evaluation of performance of devices and processes, and examining the exergy at different points of energy-conversion system. Then, using the results to quantify and qualify the losses over the whole system resulting in a detailed view of the energy losses and efficiencies of the different devices used in the system [2].

Exergy analysis knowledge is useful in explaining the changes observed in a parameter variation and in explaining the differences between the various processes. This is particularly useful when dealing with new processes where little or no experience has been gained.

Many researchers [3–6] have recommended the use of exergy analysis to aid decision making regarding the allocation of resources (capital, research and development effort, optimization, life cycle analysis, materials, etc.) in place of or in addition to energy analysis [2]. While some researchers dedicated their studies to component exergy analysis and efficiency improvement [6, 7], others focused on systems design and analysis [8–15].

The main objective of this work is to perform exergy analysis on the gas turbine power plant with two configurations, compare the exergy results with those obtained by energy analysis, and determine the primary energy loss and exergy destruction over the plant [16–19].

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2. PLANT DESCRIPTION

The simple cycle gas turbine consists of a compressor, combustion chamber, and a turbine, which is mechanically coupled to the compressor (Fig. 1). In this work the air is compressed from atmospheric pressure and delivered to two turbines arranged in two different configurations:

1. Parallel: where the combustion gases expanding to atmospheric pressure in each turbine, one of the turbines drives the compressor (CT), while the other develops the power output of the plant (PT) (Fig. 2).

2. Series: where the combustion gases expand through one of the turbines, which drives the compressor (CT), then reheated and expanded to atmospheric pressure in a free power turbine which develops the power output of the plant (Fig. 3).

Each turbine in the two configurations has its own combustion chamber; the fuel supply to each can be controlled independently.

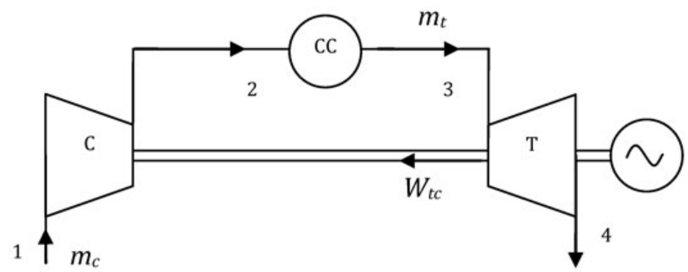

Fig. 1. Simple configuration.

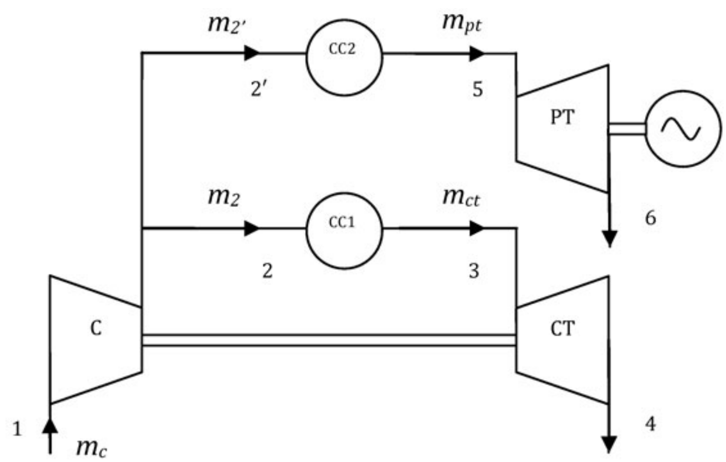

Fig. 2. Parallel configuration.

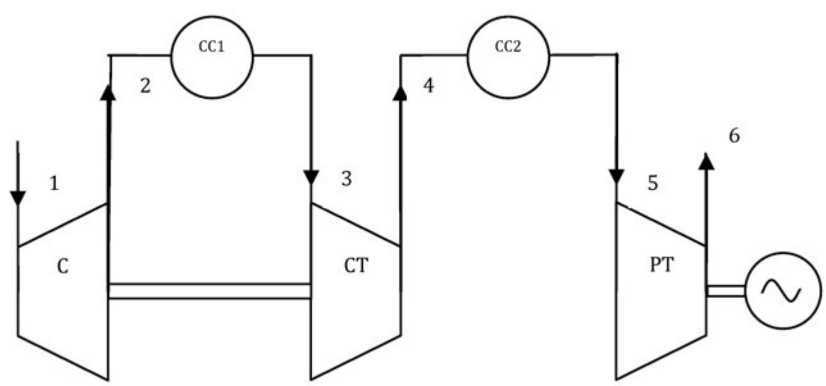

Fig. 3. Series configuration.

3. THEORETICAL ANALYSIS

3.1. Exergy Balance

The governing equations used to analyze any component in terms of exergy are as follows [16–19]:

$$\underbrace{X_{in} - X_{out}}_{\text{Net exergy transfer by heat, work, and mass}} - \underbrace{X_{dest}}_{\text{Exergy destruction}} = \underbrace{\Delta X_{system}}_{\text{Change in exergy}}.$$

For steady-state steady-flow systems (SSSFs):

$$\sum \left(1 - \frac{T_0}{T_k}\right) Q_k - W + \sum_{in}(m\psi) - \sum_{out}(m\psi) - X_{dest} = 0,$$

$$\psi = (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz,$$

$$h - h_0 = c_{p,avg}(T - T_0),$$

$$s - s_0 = c_{p,avg} \ln\left(\frac{T}{T_0}\right) - R \ln\left(\frac{p}{p_0}\right),$$

where, T_0 and p_0 are ambient temperature and pressure, respectively.

If the system is stationary, then $(V^2/2)$ and (gz) are equal to zero:

$$\psi = (h - h_0) - T_0(s - s_0).$$

Combine the equations:

$$\psi = c_{p,avg}(T - T_0) - T_0 \left[c_{p,avg} \ln\left(\frac{T}{T_0}\right) - R \ln\left(\frac{p}{p_0}\right)\right].$$

Reversible work:

$$W = W_{rev} \text{ when } X_{dest} = 0.$$

For single-stream

$$W_{rev} = m(\psi_1 - \psi_2) + \sum \left(1 - \frac{T_0}{T_k}\right) Q_k.$$

Then for adiabatic, single-stream

$$W_{rev} = m(\psi_1 - \psi_2).$$

Second-law efficiency:

For work consuming device:

$$X_{dest} = W_{in} - W_{in,rev},$$

$$\eta_{II} = \frac{W_{in,rev}}{W_{in}}.$$

For work producing device:

$$X_{dest} = W_{out,rev} - W_{out},$$

$$\eta_{II} = \frac{W_{out}}{W_{out,rev}}.$$

3.2. Theoretical Exergy Analysis of Gas Turbine Components

Applying the previous equations to the main components of gas turbine cycle, to obtain exergy destruction and second-law efficiency

Compressor (Fig. 4):

As the compressor is an adiabatic single-stream device, then:

$$W_{in,rev} = m(\psi_1 - \psi_2),$$

$$X_{dest} = W_{in} - W_{in,rev},$$

where W_{in} is the actual work supplied to compressor.

Combustion chamber (Fig. 5):

Exergy balance

$$m_1\psi_1 + m_f\psi_f - m_2\psi_2 - X_{dest} = 0,$$

$$X_{dest} = m_1\psi_1 + m_f\psi_f - m_2\psi_2,$$

$$\psi_f = \gamma_f HV,$$

where $\gamma_f = 1.06$ is the exergy factor based on the lower heating value

$$\eta_{II} = \frac{W_{out}}{W_{in}} = \frac{m_2\psi_2}{m_1\psi_1 + m_f\psi_f}.$$

Turbine (Fig. 6):

As turbine is an adiabatic single-stream device, then:

$$W_{out,rev} = m(\psi_1 - \psi_2),$$

$$X_{dest} = W_{out,rev} - W_{out},$$

where W_{out} is the actual work produced by turbine.

$$\eta_{II} = \frac{W_{out}}{W_{out,rev}}.$$

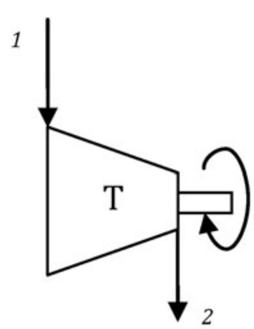

Fig. 4. Compressor schematic diagram.

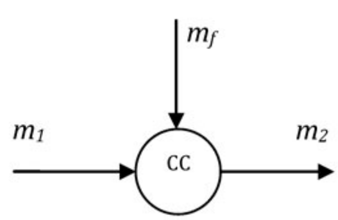

Fig. 5. Combustion chamber schematic diagram.

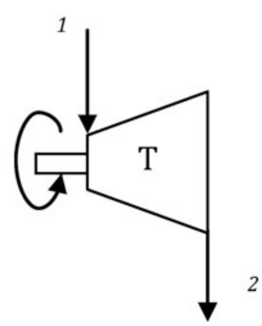

Fig. 6. Turbine schematic diagram.

4. RESULTS AND DISCUSSION

Exergy analysis is performed on the gas turbine arrangements described previously. The results of exergy analysis at design and off-design points are presented and discussed. Table 1 shows the exergy destruction of each component for each configuration at design point. Figures 7 and 8 illustrate the information shown in Table 1, while Figs. 9–12 show the exergy destruction in each component for the two configurations and the control methods.

The results show that the major component where exergy destruction occurs is the combustion chamber (above 90%). The exergy destruction in each combustion chamber depends on the quantity

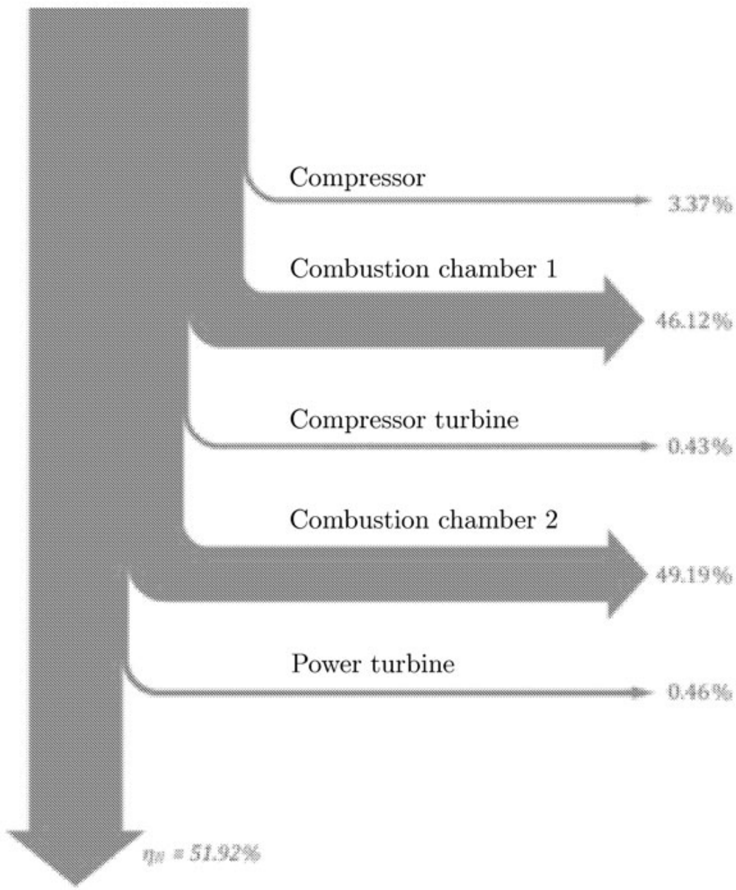

Fig. 7. Parallel exergy loss diagram.

Table 1. Exergy analysis for the two configurations at design-point

	Parallel				Series			
	X_{dest}, kW	η_{II}, %	X_{perc}, %	W_{rev}, kW	X_{dest}, kW	η_{II}, %	X_{perc}, %	W_{rev}, kW
Compressor	252	91.88	3.37	2850	252	91.88	2.91	2850
Burner 1	3444	61.28	46.12		7118	61.28	82.29	
Compressor turbine	32	98.98	0.43	3165	44	98.62	0.51	3177
Burner 2	3674	61.28	49.19		1158	89.89	13.38	
Power turbine	35	98.98	0.46	3377	48	98.74	0.55	3774
Gas turbine engine	7468	51.92			8650	54.21		

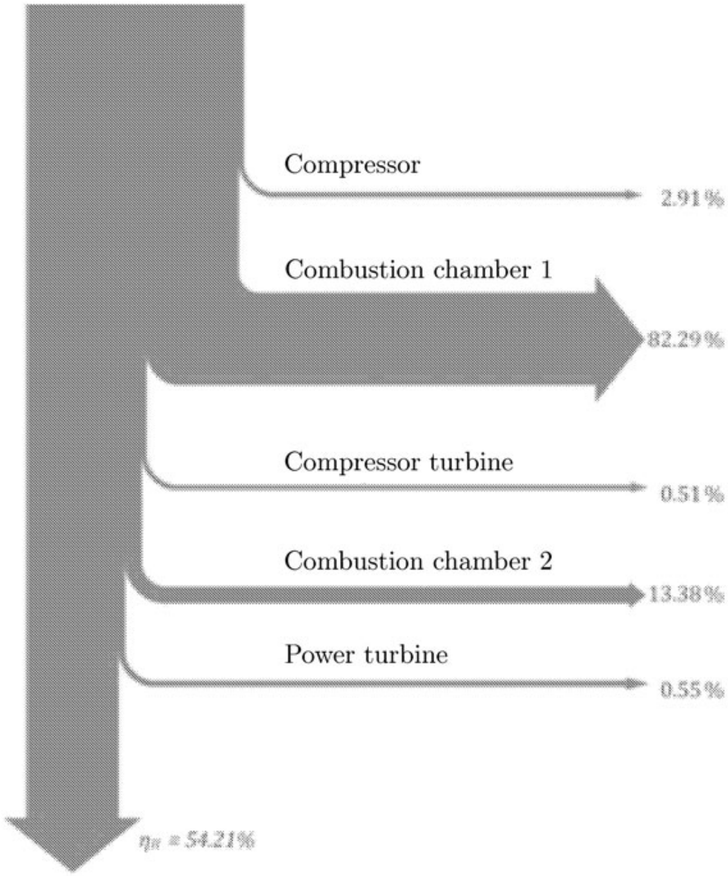

Fig. 8. Series exergy loss diagram at design point.

of the fuel consumed. This is the reason for the difference of exergy destruction between combustion chambers in the series configuration.

Table 1 with Figs. 4 and 5 shows that the parallel configuration has less second-law efficiency than the series (52% vs. 54%, respectively). This result is due to the parallel configuration having more loss sources than series. One of the major losses that exists in the parallel and doesn't exist in series is: the lost energy of the hot gases expanded to the atmosphere by two separate exhaust streams. Although the series configuration has higher second-law efficiency than the parallel configuration (by 4%), it has higher total exergy destruction (by 16%) since the series configuration involves more heat addition and work production than the parallel configuration. It is clear from Figs. 9–12 that the exergy destruction with part-load or off-design conditions depends on the variation of the thermodynamic quantities (i.e., pressure, temperature, work produced and consumed, etc.) with variation of pressure ratio, and certainly this depends on the method used to control the engine.

For series configuration when using control A ($T_3 = \text{const}$), exergy destruction of combustion chamber 1 *increases* with *decrease* of pressure ratio, while the exergy destruction across combustion chamber 2 *decreases* with the *decrease* of pressure ratio. The exergy destruction of turbo-machines (compressor and turbine) also *decreases* with *decreasing* pressure ratio.

For control B ($T_3 = T_5$) the exergy destruction in compressor, turbine, and combustion chamber 1 keeps decreasing with the decrease of pressure ratio, while increasing in the combustion chamber 2 with the decrease of compressor pressure ratio.

For parallel configuration with control A ($T_3 = \text{const}$), the exergy destruction shows a slight increase with the decrease of pressure ratio in the combustion chamber 1 and the power turbine. It shows a slight decrease with the decrease of pressure ratio in the compressor and the compressor turbine, and a severe decrease in the combustion chamber 2. For control B ($T_3 = T_5$), there is a slight decrease of exergy

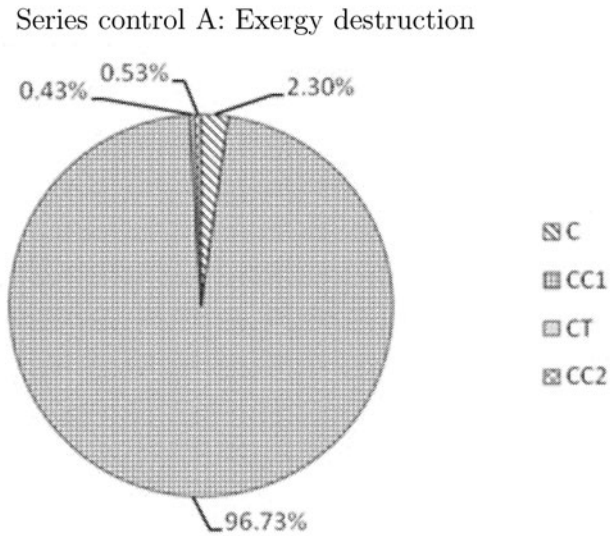

Fig. 9. Exergy destruction of the components on off-design point (80% load) in series using control A.

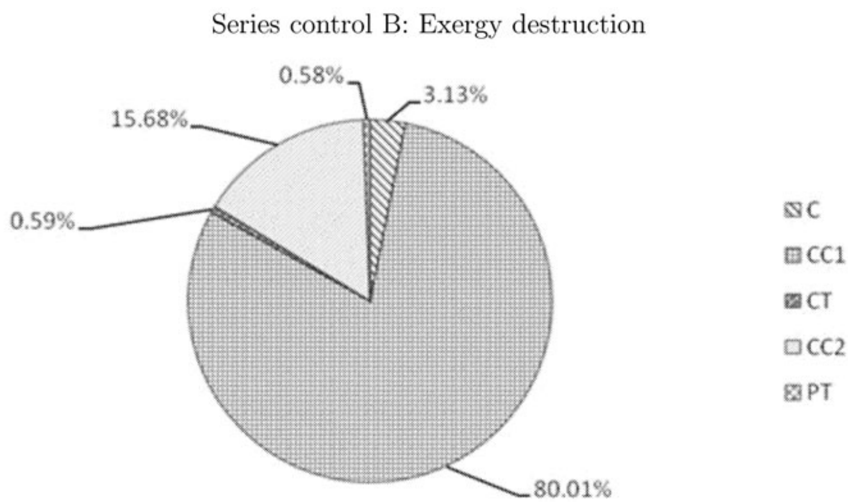

Fig. 10. Exergy destruction of the components on off-design point (80% load) in series using control B.

Table 2. Relative effect of 10% pressure ratio drop on total exergy destruction

	Parallel	Series
Control A	−8%	+1.5%, no reheat −2.5%, reheat
Control B	−3.5%	−12%

destruction with the decrease of pressure ratio in compressor, combustion chamber 1, and combustion chamber 2, but there is a slight increase with decrease of pressure ratio in both CT and PT turbines.

For off-design conditions, as the pressure ratio decreases the total exergy destruction varies differently in different configurations and methods of control, Table 2 shows the exergy destruction variation rates with decreasing pressure ratio by 10%.

Although the exergy destruction varies over off-design conditions, the second-law efficiency in most

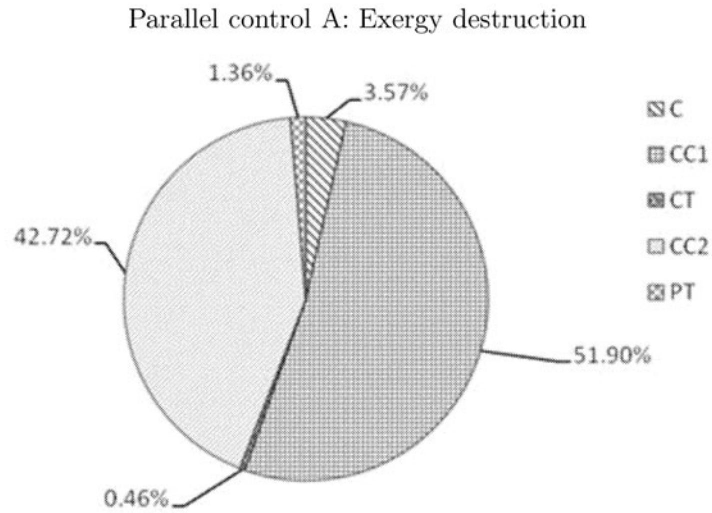

Fig. 11. Exergy destruction of the components on off-design point (80% load) in parallel using control A.

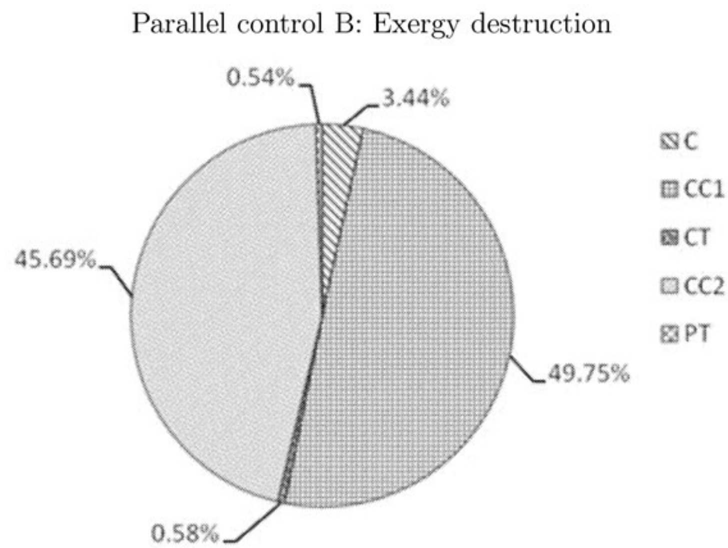

Fig. 12. Exergy destruction of the components on off-design point (80% load) in parallel using control B.

Table 3. Relative effect of 10% pressure ratio drop on second-law efficiency

	Parallel	Series
Control A	−1%	−0.1%, no reheat −1.5%, reheat
Control B	+0.1%	−7%

points and cases is not affected very much, because of the accompanied variation of available work with variations of pressure ratio; Table 3 shows the second-law efficiency variation rates with decreasing pressure ratio by 10%.

5. CONCLUSIONS

(a) The major sites where the exergy destruction occurs are the combustion chambers (above 90%), hence, efforts are encouraged to focus on the improvement of combustion process, and burner design.

(b) The parallel configuration has a lower second-law efficiency than the series (52% and 54%, respectively).

(c) The exergy destruction with part-load depends on the variation of the thermodynamic quantities (pressure, temperature, etc.) with pressure ratio, which depends on the method used to control the engine.

(d) For part-load conditions, as the pressure ratio decreases by 10%, the total exergy destruction and second-law efficiency vary differently for different configurations and methods of control.

(e) From the second-law point of view, it is recommended to use series configuration since it has relatively higher second-law efficiency than parallel.

(f) From the second-law point of view, both methods of control could be used since the second-law efficiency is slightly affected by pressure ratio variation as shown in Table 3.

NOTATIONS

m—mass flow rate, kg/s

p—pressure, bar

T—temperature, K

r—pressure ratio

c_p—specific heat of air, kJ/kg·K

W—work rate, W

HV—lower heating value of the fuel, kJ/kg

X—exergy rate, W

h—specific enthalpy, kJ/kg

s—specific entropy, kJ/kg·K

Greek Letters

η—efficiency

ψ—specific exergy, kJ/kg

Subscripts

c—compressor

t—turbine

cc—combustion chamber

0—environment

II—second-law

a—ambient, air

g—gas

rev—reversible

dest—destruction

REFERENCES

1. Rosen, M. and Dincer, I., *Exergy, Energy and Sustainable Development*, UK: Elsevier, 2007.
2. Utlu, Z. and Hepbasli, A., A Review on Analyzing and Evaluating the Energy Utilization Efficiency of Countries, *Ren. Sust. Energy Reviews*, 2007, vol. 11, pp. 1–29.
3. Szargut, D.R. and Morris, F.R. Steward, *Exergy Analysis of Thermal, Chemical, and Metallurgical Processes*, New York: Hemisphere, 1988.
4. Rosen, M. and Dincer, I., Exergy as the Confluence of Energy, Environment and Sustainable Development, *Exergy Int. J.*, 2001, vol. 1, iss. 1, pp. 1–11.
5. Dincer, I. and Cengel, Y.A., Energy, Entropy and Exergy Concepts and Their Roles in Thermal Engineering, *Entropy*, 2001, vol. 3, no. 3, pp. 116–149.
6. Song, T.W., Sohn, J.L., Kim, J.H., Kim, T.S., and Ro, S.T., Exergy-Based Performance Analysis of the Heavy-Duty Gas Turbine in Part-Load Operating Conditions, *Exergy*, 2002, vol. 2, iss. 2, pp. 105–112
7. Kwon, Y., Kwan, H., and Oh, S., Exergo-Economic Analysis of Gas Turbine Cogeneration Systems, *Exergy*, 2001, vol. 1, iss. 1, pp. 31–40.
8. Badran, O., Gas-Turbine Performance Improvements, *Appl. Energy*, 1999, no. 64, pp. 263–273.
9. Carcasci, C. and Facchini, B., Comparison between Two Gas Turbine Solutions to Increase Combined Power Plant Efficiency, *Energy Conv. Man.*, 2000, vol. 41, pp. 757–773.
10. Verkhivker, G. and Kosoy, B., On the Exergy Analysis of Power Plants, *Energy Conv. Man.*, 2001, vol. 42, pp. 2053–2059.
11. Marrero, I., Lefsaker, A., Razani, A., and Kim, K., Second Law Analysis and Optimization of a Combined Triple Power Cycle, *Energy Conv. Man.*, 2002, vol. 43, pp. 557–573.
12. Sue, D. and Chaung, C., Engineering Design and Energy for Combustion Gas Turbine Based Power Generation System, *Energy*, 2004, vol. 29, pp. 1183–1205.
13. Aljundi, I.H., Energy and Exergy Analysis of a Steam Power Plant in Jordan, *Appl. Thermal Eng.*, 2009, vol. 29, pp. 324–328.
14. Kotas, T.J., *The Exergy Method of Thermal Plant Analysis*, US: Krieger, 1995.
15. Cengel, Y.A. and Boles, M.A., *Thermodynamics: An Engineering Approach*, 7th ed., US: Mc-Graw Hill, 2007.
16. Khaldi, F. and Adouane, B., Energy and Exergy Analysis of a Gas Turbine Power Plant Algeria, *Int. J. Exergy*, 10.1504/IJEX.2011.043920.
17. Nieto, R., Gonzalez, C., Lopez, J., and Jimenez, A., Efficiency of a Standard Gas-Turbine Power Generation Cycle Running on Different Fuels, *Int. J. Exergy*, 10.1504/IJEX.2011.041433.
18. Alasfour, F.N., Al-Fahed, S.F., and Abdulrahim, H.K., The Effect of Elevated Inlet Air Temperature and Relative Humidity on Gas Turbine Cogeneration System: Exergy Assessment, *Int. J. Exergy*, 10.1504/IJEX.2011.039789.
19. Farshad, A., Shams, M., and Behbahani-nia, A., Sensitivity Analysis and Thermo-Economic Optimization of Steam Injected Gas Turbine Cycle Design Parameters, *Int. J. Exergy*, 10.1504/IJEX.2011.039791.